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From: Chief, Naval Technical Mission to Japan.
To : Chief of Naval Operations.

Subject: Target Report - Characteristics of Japanese Naval Vessels, Article 11 - Main and Auxiliary Machinery.

Reference: (a)"Intelligence Targets Japan" (DNI) of 4 Sept. 1945.

1. Subject report, covering a portion of Targets S-01 and S-05 of Fascicle S-1 of reference (a), is submitted herewith.

2. The investigation of the target and target report were accomplished by Mr. J. A. Davies, Civilian Technician, assisted by Lt. (jg) G. Sheeks, USNR, interpreter.

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RESTRICTED S-01-11

CHARACTERISTICS OF JAPANESE NAVAL VESSELS
ARTICLE 11
MAIN AND AUXILIARY MACHINERY

"INTELLIGENCE TARGETS JAPAN" (DNI) OF 4 SEPT. 1945

FASCICLE S-1, TARGETS S-01 AND S-05

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U.S. NAVAL TECHNICAL MISSION TO JAPAN

SUMMARY

SHIP AND RELATED TARGETS

CHARACTERISTICS OF JAPANESE NAVAL VESSELS - ARTICLE 11
MAIN AND AUXILIARY MACHINERY

A review has been made of a number of blueprints of assemblies and details of the propulsion and auxiliary machinery for Japanese naval vessels.

In addition, a statement of the latest design factors has been obtained from the Fifth Section (Machinery Design) of the Technical Department of the Japanese Navy Ministry.

Except perhaps for the compact arrangement of the cylinder and nozzle block, the design of auxiliary turbines and gears does not present any advancement that is worthy of more than passing attention. Certain features such as separate pinion sleeves have been tried before in the United States and have not proved satisfactory. The general design requires more production man-hours and is more costly than are the types that are presently available in the U.S.

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INTRODUCTION

The working drawings of turbines and gears as applied to a battleship, an aircraft carrier and a destroyer were reviewed to ascertain whether any outstanding contribution had been made by engineers of the Japanese Navy, particularly during the past few years.

Subsequently, one of the leading Japanese machinery designers was interrogated to learn whether the drawings reviewed exhibited the latest practice.

THE REPORT

Part I - MAIN TURBINES

Aircraft Carrier KATSURAGI - The drawings for the main propulsion machinery of KATSURAGI were completed in 1943 and thus can be regarded as representative of the latest design practice for Japanese naval vessels. The designed shp per ship is 104,000 divided among four shafts. The designed RPM of the prepellers is 340.

The power transmitted to each shaft at the designed full speed is divided among three turbines, HP, IP, and LP, each driving an independent pinion on the main gear wheel.

The HP machine is at the after end of the main reduction gear while the IP and LP units are placed forward of the gear.

All three turbines are of the impulse type. The HP has a two-row Curtis wheel followed by two Rateau stages. The IP has five Rateau stages and the LP is a double flow machine with two two-row wheels on each side of the vertical center line. The single astern turbine is located at the forward end of the LP unit and consists of a three-row, single stage wheel.

A cross-compound cruising turbine with a two-pinion single reduction gear connects to the forward end of the IP turbine rotor. The connection is made through a disengaging jaw type clutch fitted with a mechanical synchronizing device.

The main turbine rotors are of the built-up type with separate discs pressed on to a shaft. Apparently the steel forging industry of Japan has not been able to handle solid forgings of the size required or else the designers were dubious of the soundness of the big forgings necessary. The latter is the more plausible explanation. According to information received at KURE there existed a forging plant with ample capacity in presses and other necessary equipment to make forgings of the size required for these turbine rotors. The cruising turbine rotors on this design were made from solid forgings.

In these latest designs no attempt has been made to use fabricated and welded steel plate for such parts as the LP cylinder or gear wheels and casings. Single reduction gears persist and as a consequence the turbine speeds are much below those possible when double reduction gears are used. The size and weight increases accordingly. Enclosure (A) contains the results of an interview with representatives of the Japanese Navy giving further information regarding the technical thinking of that organization.

The details of design of both the turbines and gears follow a conventional pattern.

Part II - REDUCTION GEARS

AKITSUKI Class Destroyer Gears - Each propeller shaft is driven by a three-pinion reduction gear transmitting a total of 26,000 shp at 343 RPM, the HP and IP units being in the outboard side and the LP on the inboard side. The HP turbine is aft of the gear and the IP and LP forward. At the forward end of the IP turbine is a clutched geared cruising unit having two pinions driven by a cross-compound turbine.

Both the cruising and main gears are single reduction and follow the conventional pattern that existed in the United States some fifteen to twenty years ago.

The K factor of the gears at designed full power is 103 on the main and 101 on the cruising gears. This and other pertinent data of the two units are given in Table I.

The pinions are of forged nickel-steel, oil quenched with a Brinnell hardness of 200 minimum. The rims are of forged carbon steel with integral flanges on the inner diameter for bolting to the side plates which connect the rim to the shaft. This latter has integral bolting flanges for the side plates. Thus the forging for both the rims and shafts contain a great deal of material which must subsequently be machined off, making the machine shop operations lengthy and costly. To this must be added the time required for drilling and bolting the parts together. Present U.S. Navy practice of making an all-welded wheel structure would seem to be distinctly superior, not only in time of production but also in cost.

The shaft stress at full load, due only to torsion is 8000 psi. The shearing stress on the coupling bolts is 6300 psi, but, on the after row of bolts holding the side plate to the shaft, the shearing stress is 9300 psi, neglecting friction in each case. The bolts holding the side plates to the rim and the shaft are riveted over the nut to prevent loosening.

The teeth have a full fillet at the root, and tip relief is provided by the hob which has an angle of obliquity of $14\frac{1}{2}^{\circ}$. The ends of the gear rim teeth are chamfered in the usual way as are also the pinion teeth, but the latter have additional relief on the flanks at the ends of the teeth extending in for about 3/8", a practice common on English gears and occasionally used in the United States.

The couplings connecting the turbine to the pinions are of the jaw type similar to those used on the machinery for the United States Navy DD 397-9 class and earlier. Lubrication of the working faces is provided by oil holes leading to the bottom of the space between the jaws; the oil to these holes is supplied from a centrifugal lubrication well formed in the end of the coupling hub. No oilways are cut in the working surfaces of the coupling jaws. The total end-play provided by the coupling is 13/32". The coupling bolts are secured from backing off by the use of castle nuts.

The lower half of the gear case is of cast steel while the upper half is fabricated and welded of steel plate. The thickness of the walls of the steel casting is generally one inch while the steel plate in the cover is 3/8". The joint flanges are $1\frac{1}{2}$" thick and are secured together with 1" and 1 1/8" bolts. The oil pan is of welded steel, 1/8" thick with flanges 5/8" thick secured to the lower gear case by 5/8" tap bolts.

The propeller thrust bearing is not a part of the reduction gear.

A worm and wheel type of turning gear is fitted to the after end of the LP pinion.

The gear case is fitted with air vents of the mushroom type designed to prevent the escape of oily vapors. It is similar in construction to those formerly used in American gears and probably is none too effective in preventing an oily smear in its vicinity on the top of the gear case. Two such vents are shown on each gear case.

Table I
AKITSUKI CLASS DESTROYER (#365)
GEAR DATA

Item	Main Gear				Cruising Gear			Astern
	HP	IP	LP	Gear Wheel	HP	IP	Gear Wheel	
SHP	8000	8750	9250	26000	1600	1900	3900	10000
RPM	3618	3002	2613	343	7750	5460	1630	200
No. Teeth	39	47	54	415	31	44	147	
Ratio of Reduction	10.64	8.83	7.69		4.74	3.34		
Diametral Pitch	5	5	5	5	7	7	7	
P.C.D. (in)	8.9	10.33	12.25	94.5	4.92	7.00	$23\frac{1}{2}$	
Length Face (in)	$38\frac{1}{2}$	$38\frac{1}{2}$	$38\frac{1}{2}$	$38\frac{1}{2}$	$12\frac{1}{2}$	$12\frac{1}{2}$	$12\frac{1}{2}$	
Helix Angle	30°	30°	30°	30°	30°	30°	30°	
Tooth Press (lbs)	31500	35600	36500		5140	6260		
K Factor	103	100	88		101	93		
No. Bearings	3	3	3	2	2	2	2	
Bearing Diameter (in)	7	7	$8\frac{1}{2}$	15	3 3/8	4 1/8	4 3/4	
Bearing Length — Center	15x1	15x1	15x1					
Bearing Length — Ends (in)	$8\frac{1}{2}$x2	$8\frac{1}{2}$x2	$8\frac{1}{2}$x2	$16\frac{1}{2}$x2	$5\frac{1}{2}$x2	$5\frac{1}{2}$x2	$5\frac{1}{2}$x2	
Journal Speed fps	110	90	97	24	112	98	34	
Bearing Press psi	140	158	134	125	148	150		
Ratio- L/D Each Helix	2.2	1.9	1.6		1.27			

The diametral pitch is to the nearest whole figure and is taken in the plane of rotation. The estimated weight of the cruising gear is 5500 lbs and of the main gear 76,000 lbs.

Table II
AIRCRAFT CARRIER KATSURAGI
REDUCTION GEAR DATA

(4 shafts-32.00 knots)

Part	Main Gear Pinions				Cruising Gear		
	HP	IP	LP	Wheel	HP	IP	Wheel
SHP	8000	8750	9250	26000			
RPM	3618	3003	2613	340	10216	7198	2154
No. Teeth	39	47	54	415	31	44	147
Diametral Pitch	5	5	5	5	7	7	7
Helix Angle	30°	30°	30°	30°	30°	30°	30°
P.C.D.	8.9	10.6	12.2	94.1	4.9	7	23½
Length of Face (in)	38½	38½	38½	38½	12 5/8	12 5/8	12 5/8
Ratio L/D (incl. 2 3/8 gap)	4.32	3.63	3.15		2.58 3.05	1.80 2.15	
Ratio of Reduction	10.6	8.8	7.7		4.75	3.35	
Total Tooth Press	31500	35600	36500				
K	103	100	88				
Bearings No.	3	3	3	2	2	2	2
Bearing Diameter (in)	7	7 3/4	8 5/8	16	3 3/8	4 1/8	4 3/4
Bearing Length (in)	2-8 5/8 1-15	2-8 5/8 1-15	2-8 5/8 1-15	2-17¼	5½x2	5½x2	5½x2
Bearing Press psi	140	158	134				
Bearing Velocity fps	110	90	9.7		150		

Note: It is evident that the reduction gears on this ship are of the same design as those on the Japanese DD, AKIT-SUKI. The division of power on the cruising unit has not been obtained.

Part III - AUXILIARY TURBINES AND GEARS

The Japanese Navy developed three standard turbo-gear drives for auxiliary units: 120, 180 and 270mm machines, the number referring to the mean diameter of the blades on the single wheel of the turbine. Expressed in inches these are 4 3/4, 7 1/8, and 10 3/8 diameter, respectively.

The detail construction of the two smaller units is similar, while the 270mm size follows another pattern of turbine rotor construction.

The description given is therefore confined to one of each size, the 180 and 270mm. (See Table III.)

Two types of a larger machine, 360mm were developed. That shown on NavTechJap Document No. ND50-1909.8 (see Enclosure (D)) was made for test purposes only and only one was built. It incorporates a thrust collar feature on the pinion that was copied from a Brown Boveri design. Of the other design, shown on NavTechJap Document No. ND50-1909.7, four were built to drive the main feed pumps on the destroyers AMATSUKAZE and SHIMAKAZE.

The design of these 360mm units parallels that of the 270mm machine so that the description given of the latter machine will suffice. The designed horsepower and maximum RPM for the three standard frames are as follows:

120mm	30hp	19,000 RPM
180mm	20-50hp	18,000 RPM
270mm	100-400hp	12,000 RPM

The designed steam conditions were 370 psi and 650°F. with atmospheric back pressure. For the 360 RPM units used on the destroyers the pressures and temperature were 570 psi and 850°F.

The turbines and gears are the latest design of such equipment. The drawings bear an original date, as given in the following table, and prints of the assemblies that have been obtained bear renewal dates of the year 1942 (See Enclosure (D)).

	Drawing	Year
Turbine and Gear for Main Feed Pump.	E 311-5	1939
Turbine and Gear for Blower.	3271-1	1937
Turbine and Gear for Circulating Pumps.	3238	1937
Turbine and Gear for Condensate.	E 184-5	1939
Turbine and Gear for Fuel Oil.	E 3113-1	1938

The turbines for all of these have a single stage, two-row Curtis wheel and operate non-condensing.

The gears are single helical with Maag type teeth. The circulating pump has a double reduction gear. All of the others have single reduction.

The turbine rotors are overhung from the pinion shaft bearings.

<u>180mm Rotor</u>. The disc which carries the second row of rotating blades is forged solid with a shaft on which a separate pinion sleeve is pressed and keyed. The first row of blades is carried on a separate disc which is also pressed on and keyed to the same shaft. The turbine blades have the pinned type of fastening, the blade roots straddling the disc. The reason for making one disc loose is evidently to permit riveting the blade pins as there is only about 11/16" space between the two rows of blades when the rotor is completely assembled.

<u>270mm Rotor</u>. The disc is forged solid with the pinion shaft and carries both rows of blades which have roots of the conventional tee section.

Turbine thrust. At the end of the shaft remote from the turbine rotor, a single collar thrust is provided. The babbitt faced disc which carries the major load has a spherically shaped back to permit a degree of self-adjustment.

Pinions. All of the pinions are loose sleeves with a tapered bore to be pressed and keyed to the turbine rotor extension shaft. It is assumed that the teeth are cut before the sleeve is mounted on the shaft.

Gear wheels. These are made from disc forgings which are pressed and keyed on to a taper turn on the gear shaft, being located in place by a nut.

Bearings. There are two bearings to each shaft. They are supported in the gear housing. The gear case cover must be lifted to get at the bearings.

Lubrication. On the thrust end of the pinion shaft for the condensate pump, a small centrifugal pump runner has been attached, presumably to act as a lubricating pump since the section pipe appears to lead to the bottom of the gear case sump. This feature is not indicated on the other drives. The circulating pump has a gear type oil pump driven off the slow speed gear shaft.

Governor. Each unit has a shaft type auto stop governor as a protection against overspeeding. This is mounted in the slow speed gear shaft.

Tachometer. On the casing at the end of the slow speed gear shaft a place is provided for mounting a tachometer. The unit for the fuel oil pump has a permanent attachment between the gear shaft and the tachometer spindle. On the others the tachometer spindle is held away from the gear shaft by a spring. An external lever is provided to permit the operator to bring the tachometer spindle into contact with the rotating shaft whenever a reading of speed is required.

Table III
AUXILIARY TURBINES AND GEARS

	SHP	RPM		Mean Dia. Turb. Wheel		Steam				Exhaust Press.		Nozzles				
		Turbine	Pump	mm	in	Pressure		Temperature		kgs/cm^2	p.s.i.	No.	Throat Dia.		Total Area	
						kgs/cm	p.s.i.	°C	°F				mm	in	cm	sq in
Feed Pump	320	12388	4000	270	10 5/8	26	370	335	630	1	15	14	6	.236	3.962	.615
Blower	200	12444	3000	270	10 5/8	26	370	335	630	1		8	6	.236	2.262	.35
Circulating Pump	220	12200	3110 650	270	10 5/8	26	370	335	630	1		9	6	.236	2.544	.395
Condensate Pump	22	12000	1900	180	7 1/8	26	370	335	630	1		2	4.8	.189	.36	.056
Fuel Pump	35	11370	180	180	7 1/8	26	370	335	630	1		1	5.7	.225	.258	.04

	P.C.D.				No. Teeth		Helical Angle	Width of Face		Ratio of Reduction	Module	Diametral Pitch - Hob in	K
	Pinion		Gear										
	mm	in	mm	in	Pinion	Gear		mm	in				
Feed Pump	112.28	4.40	347.72	13.60	31	96	13° 30'	75	2.95	3.1	3.5	7	72
Blower	101.01	4.00	418.99	16.50	27	112	19° 50'	75	2.95	4.15	3.5	7	74
Circulating Pump	97.5 117.62	3.83 4.70	382.5 562.38	15.00 22.30	26 32	153	19° 60' 17° 8'		2 3/8 4 5/8	3.93 4.80	3.5	7	104
Condensate Pump	60.42	2.37	381.58	15.00	19	120	17° 7½'	35	1 3/8	6.32	3	8	35
Fuel Pump	50.36	2.00	381.64	15.00	19	144	17° 20'	50	1.96	7.58	2.5	10	55

ENCLOSURE (A)

REPORT ON THE LATEST STANDARDS OF DESIGNS FOR THE PROPULSION MACHINERY OF THE JAPANESE NAVY

* * * * *

SUMMARY

A review of the many blueprints which had been collected failed to disclose any outstanding feature of design. The information on the drawings with respect to materials was incomplete and none of the papers available gave any upper limits to stresses or steam conditions which the Japanese designers may have established.

An interview was arranged for 8 February 1946 at the NISSON Building which housed the Technical Department of the Navy Ministry.

Those present were:

For United States Navy,

Mr. J.S. Davies, Civilian Technician
Lt. (jg) G. Sheeks, Interpreter.

For the ex-Imperial Japanese Navy,

Commander K. KUDO
Technical Captain M. YASUGI

The interview began at 0930 and concluded at 1200.

A series of questions had been prepared by the U.S. representative and these were presented to the proper authorities in the NISSON Building two days prior to the meeting. Both the questions and the answers are given on the following page.

* * * * *

ENCLOSURE (A), continued

TURBINES

1. Q. Why was the reaction type (Parsons type) abandoned in favor of straight impulse?

   A. Blade breakage. The Japanese persisted in the use of naval brass and and phosphor bronze blade material with lashing wires in holes in the edges of the blades. This type of turbine was last used on the cruisers FURUKATE and AOBE. Five years ago the IJN purchased from Brown Boveri two 5000 shp reaction turbines which were installed on a transport. These machines had stainless steel blades with lashing wires in the center of the blade section. They were in service about a year before the ship was sunk but in that time gave no blade trouble. No attempt was made to develop a better blade construction.

2. Q. Why were built-up rotors used instead of solid forgings?

   A. A lack of the technical ability to produce such large forgings. The original LP spindles for NAGATO were solid but one of them burst on test and this experience caused the Navy to prohibit the use of solid forged turbine rotors except in the smaller sizes for auxiliaries and cruising units.

3. Q. Was any attempt made to use cobalt chromium (Stellite) seats for steam valves?

   A. No. Material was not available. Stainless steel inserts of different hardnesses were used. The first seats were made with both valve and seat inserts of the same hardness with the result that seizure and galling took place.

4. Q. Was any trouble experienced with thrust bearings on turbines, particularly generator turbines, due to water coming over with the steam? How was this problem overcome?

   A. Yes, plenty. All of the generator turbines and the cruising turbines were fitted with a Michel type thrust on the maximum pressure side and a plain collar shoe on the opposite side. The thrust was found to reverse and in every case the babbitt face of the flat collar burned out. The thrust was changed on all turbines to a double Michel type and the surface finish of the thrust collar was improved. Since then very little trouble has been experienced.

   Steam separators are fitted in the steam lines to generators but they are not regarded as a complete assurance that slugs of water will not reach the turbine.

5. Q. What difficulty has been found with leakage at the turbine joints and how was this overcome? How are joints prepared?

   A. Occasional leaks have been experienced particularly at the intersection of the vertical and horizontal joints. This has led to the staggering of the vertical and horizontal flanges. Manganisite is used between cylinder joints. No other preparation such as triple boiled linseed oil or usudurian has been tried.

6. Q. Have there been any serious erosion problems on turbine blades?

   A. Not on the stainless steel blades. Previously, when nickel steel was used considerable erosion took place. No attempt is made to entrap water to lessen erosion of the last row of LP blades.

ENCLOSURE (A), continued

7. Q. Have any blade failures been experienced due to resonant vibration? How was the vibration induced? Were any traceable to periodic error in the gears?

A. Yes, several. The most troublesome was a second mode vibration on a complete admission IP wheel. The number of the vanes of the nozzles was changed.

No troubles were reported that could be charged against the gears.

8. Q. What trouble, if any, has been experienced with oil leaks past turbine bearing oil rings? Has water from the glands been found in the oil system?

A. Plenty of both. The venting of the turbine pedestals and gear cases has been improved and a large gland exhaustion fitted to a modified gland.

9. Q. What difficulties with steel castings and what procedure for repair? What about re-annealing after welding?

A. The usual shrinkage cracks occurred, sand inclusions etc. which were chipped out and welded in the regular manner. Re-annealing is required by the drawings but the shop people appeared to use their judgment in such matters.

10. Q. What bolt stresses are used for turbine joints and how are they determined?

A. Nothing of the sort has been attempted. The tightness of the bolts is left to the judgment of the shops.

11. Q. What thought has been given to higher steam conditions? What is considered the standard vacuum for naval vessels? For merchant ships?

A. The Navy has been very conservative in the matter of pressure and temperature. The highest steam conditions so far used were on YAMATO which had a pressure of 570 psi (40 kg/cm^2) at 750°F. (400°C.).

For naval vessels the turbine designers used a vacuum of 27.56 inches (700mm of mercury) and for merchant vessels 28.35 inches (720mm), both at the contract full power. The circulating water inlet temperature is taken as 77°F(25°C) and 75°F(24°C) respectively.

12. Q. What has been done with respect to extracting steam from the turbines for feed heating? Has the reheat cycle been attempted on either naval or merchant ships?

A. No, for naval vessels. A few merchant ships have used extracted steam for feed heating. The reheat cycle has not been used.

13. Q. Why is cast iron used for bearing shells and why are turbine bearings not adjustable?

A. We must have mistaken the drawings. For naval vessels, bronze was used until the supply of copper made it necessary to change and then either forged or cast steel was adopted. For merchant ships cast iron is used, while the Brown Boveri turbines (mentioned in Question 1) had adjustable bearings. All of the Japanese designs have the straight cylindrical sleeve type.

ENCLOSURE (A), continued

14. Q. What has been the experience with carbon glands?

A. Fairly satisfactory. Some trouble from vibration and an occasional grooved shaft has occurred due to insufficient clearance but not enough trouble to warrant radical change.

15. Q. What tests have geen made when operating astern? Have temperatures in the ahead blading been recorded and have casualties occurred? Where is the hottest zone?

A. No tests have been made. There have been severe casualties on LP turbines apparently from cylinder distortion on the later type impulse machines.

16. Q. Why was the straight Rateau turbine abandoned on the latest LP units in favor of the two-row Curtis element?

A. This question is due to a mistake in reading the drawing. The straight Rateau design has been retained but the nozzles are carried in the cylinder wall instead of in a diaphragm.

17. Q. Has any design been prepared with respect to putting an astern element at each end of the LP?

A. No.

18. Q. What steam velocities are regarded as acceptable in main steam lines? What is regarded as a tolerable drop in pressure between the boiler outlet and the turbine inlet?

A. With superheated steam, 164 fps (50 m/s), with saturated steam, 148 ft/s (45m/s).

Pressure drop from superheater outlet to turbine steam chest, 70 to 85 psi (5 to 6 kg/cm^2).

19. Q. What exhaust velocities in the LP turbine are permitted?

A. 400 fps (120 m/s).

20. Q. Has any attempt been made to provide axial freedom for the expansion of the turbine by the use of flexible "I" beams?

A. No.

21. Q. What protection is provided for the portion of the shaft under the carbon rings? Has chrome plating been used?

A. None. No.

22. Q. Where the exhaust from the LP turbine comes out of the top of the cylinder, as it is on some of the recent designs, what provision is made for drainage?

A. A separate small condenser and air ejector discharging to main condenser is provided.

ENCLOSURE (A), continued

GENERAL

23. Q. Was lead base babbitt used in any bearings and what was the experience with it?

A. None used.

24. Q. What shop tests were undertaken and how extensive, for main propulsion units?

A. Full power tests were conducted on two units, one of 9000 shp for a merchant type and the other a 25,000 shp naval equipment. Water brakes designed and built in Japan were used. All other turbines are tested at no load and at 20% overspeed.

25. Q. Were water brakes used and what kind?

A. See answer to Q. 24.

26. Q. Were geared turbines used for merchant ships and what kind?

A. **Turbines with double reduction gears were built at the Kawasaki Dockyard, KOBE and installed on IZUMA MARU which was later converted into a carrier and renamed HIYO. MITSUBISHI at NAGASAKI built similar machinery for KASHIWABARA MARU, also subsequently converted and named HAYATAKA. The ships were twin screw with four turbines per shaft - HP, 1st IP, 2nd IP, and LP.**

These two ships were twin screw with 28,125 shp per shaft at 170 RPM of propeller. The astern power was 13,500 shp per shaft at 144 RPM. Steam was supplied at 35 kg/cm^2 and at 405^{o}C (500 psi and 760^{o}F) at the turbine inlet. The HP and the first IP turbines operated at 4145 RPM, the second IP at 3525 RPM and the LP at 2392 RPM. The arrangement of the machinery is given on the accompanying sketch (Fig. 1 (A)).

27. Q. What type machines are used for balancing rotors, and are gear parts balanced?

A. Flexible type "I" beams. Gear parts are not balanced. An attempt was made to balance pinions but as the shops were unsuccessful, it was abandoned.

28. Q. What is the type of thrust bearing for turbines and propeller, Michel or Kingsbury or a Japanese version of both?

A. Michel for propellers. Japanese design for turbines to avoid patent royalties.

29. Q. What surface finishes are required for journals and the faces of thrust collars? How obtained?

A. Surface finish was not measured. No knowledge of profitometer. Journals were ground and thrust faces lapped.

30. Q. Has any study been made leading to the use of double reduction gears and higher speed turbines?

A. Not for naval vessels. (See 26 A. for merchant ships)

ENCLOSURE (A), continued

31. Q. Provide full particulars of material composition of chemical and physical properties used for main parts of turbines and gears.

A. (See table at end of interrogation).

32. Q. Why has the use of steel castings persisted in view of the possibilities of weight saving by the adoption of fabrication from steel plates, particularly for the LP turbine and reduction gear casings and gear wheels?

A. Gear casings are now designed for welding but the Navy has not permitted this for turbine cylinders.

Some years ago two destroyer gears were purchased in the United States and, after the full power sea trials, a crack was observed in the weld attaching the side plate to one of the rims. The gear wheels were removed and replaced by a standard built-up design at the demand of the Navy engineers. The rejected gear was put under an extensive full power test ashore without any trouble developing. The use of fabricating and welding has been retarded by the undue conservatism of the Navy.

33. Q. What are regarded as optimum bearing pressures and peripheral speeds of journals? Is any difference considered necessary between the high and low speed elements?

A. Turbines, 142 psi (10 kg/cm^2). Pinions, 170 psi (12 kg/cm^2) No. Speed for both is 148 ft/s (45 m/s).

34. Q. Has any attempt been made or considered with respect to driving any of the auxiliaries from the main units, e.g., lubrication oil pumps? What precautions are in vogue for preventing loss of oil to the propulsion turbine and gears?

A. Recently, on the destroyer SHIMEKAZE an oil pump was driven from the slow speed gear shaft. The practice is common in the merchant marine.

Two pumps are always in operation so that if one fails no reliance need be had on pressure controlled valves.

REDUCTION GEARS

35. Q. What is the history of gear tooth wear, e.g., pitting, galling and wire edges? Are all teeth, pinions and wheels, tip relieved?

A. We have had all of the troubles to which gears are heir. Galling and excessive wear have not been troublesome. Pitting is persistent but does not cause concern. All teeth are tip relieved by the hob.

36. Q. What means are used to check accuracy of hobbing machines and hobs?

A. No method such as a split ring test gear. Teeth are checked after cutting by some apparatus purchased 10 years ago from David Brown and Co. of England. A new gear hobbler bought from Reineke of Germany was used to cut the gears of YAMATO.

37. Q. Are other means used for cutting gears, such as the Sukes method of shaping teeth?

A. No.

ENCLOSURE (A), continued

38. Q. Why was the propeller thrust bearing not incorporated in the forward end of the gear housing?

A. It was not regarded as desirable for naval vessels. It is in the gear case on merchant vessels but is aft of the gear.

39. Q. What noise problems with gears have been experienced and how were they overcome? How analyzed?

A. Not a great deal of trouble with noise, particularly on the latest gears. A sound meter was used and readings of 90 decibles were recorded on a number of gears.

40. Q. What is regarded as the limit to the unit tooth pressure of gears? Is the effect of reduction ratio considered in establishing the limit?

A. $\frac{P}{\sqrt{d}}$ = 30 kg/cm for pinions with d larger than 25 cm and $\frac{P}{\sqrt{d}}$ = 6 kg/cm for smaller pinions. No consideration is given to the reduction ratio.

41. Q. Was the "Fast" type of toothed coupling ever considered for connecting turbines to pinions? In other words, why was the jaw type persisted in?

A. Brown Boveri turbine (see 1 A.) had a "Fast" type coupling. Japan tried to produce such couplings but the war prevented the purchase or development of a machine for cutting the internal gears.

42. Q. Is any attempt made to supply the gear teeth with oil or higher viscosity than that for the turbine bearings?

A. No.

43. Q. What method is used to get good contact on gear teeth if on assembly the bearing is not considered satisfactory? Are the gears ground together and, if so, how extensively and in what manner?

A. A slight scraping to the babbitt is the customary procedure. Gears cut on the old machines have such a poor finish that hand scraping of the teeth is not uncommon. Gears from the new machine, YAMATO'S for instance, were very good and required very little adjustment to get a good tooth marking.

Grinding together with an abrasive such as glass powder was abandoned some years ago.

ENCLOSURE (A), continued

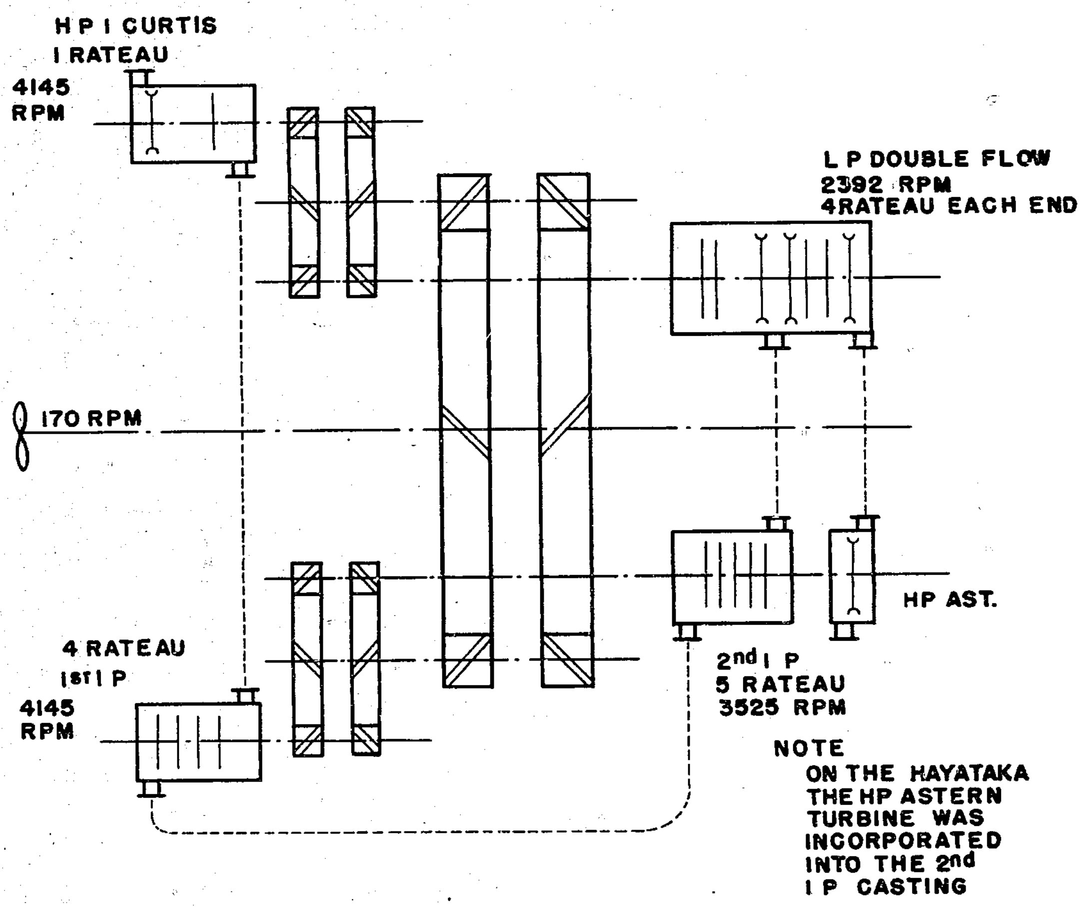

Figure 1(A)

SCHEMATIC ARRANGEMENT OF MACHINERY ON TWIN SCREW CVE HITAKA AND HAYATAKA

ENCLOSURE (A), continued

CHEMICAL PROPERTIES OF TURBINE AND GEAR PARTS

Part	No.	Use*	Composition							
			Ni	Cr	C	Si	Mn	P	S	Mo
Blade	1	A	2 max	11-15	.2 max	.6 max	.5 max	.035	.035	
	2	B	1 max	11-15	.15 max	.6 max	.5 max	.035	.035	.25-.40
Disc	1	C	Acid Furnace Steel					.055	.05	
			Basic Furnace Steel					.045	.05	
	2	A	.5-1.5	.5 max	Acid Furnace Steel			.055	.05	
					Basic Furnace Steel			.045	.05	
	3	D	3.4	.5-1	.25-.4	.35 max	.3-.8	.035	.035	
	4	B	1-2.5	.3-.9	.25-.4	.35 max	.35-.65	.035	.035	
Shaft	Same as Disc 1.									
Pinion	1	A	3.5-4.5		.25-.35	.35 max	.3-.8	.035	.035	
	2	D	3-4	.5-1	.25-.4	.35 max	.35-.65	.035	.035	
	3	C		1-1.5	.27-.37	.35 max	.3-.6	.035	.035	.2-.3
Rims	Acid Furnace Steel							.055	.05	
	Basic Furnace Steel							.045	.05	
Gear Wheel for Auxiliaries			3-4	.4-.7	.1-.5	.2	.4	.03	.03	
Turbine	1	B	Cast Steel					.02	.015	.25-.4
Cylinders	2	A	Acid Furnaces Cast Steel					.015	.06	
			Basic Furnaces Cast Steel					.055	.06	
Nozzles	1	A	4.5-5.5		.08	.3	.6	.03	.035	
	2	C			.12	.3	1.3-1.8	.035	.035	
Cast Iron	1				.12	.3		.2	.07	

* A-In general use

B-For high temperatures, 660°F and higher

C-For war time use

D-For auxiliary turbines

ENCLOSURE (A), continued

Part	No.	Use	Properties								
			Yield Point		Tensile Strength		Elong (%)	Red (%)	Izod Value	Brinell Hardness No.	Bend
			(kg/mm^2)	(psi)	(kg/mm^2)	(psi)					
Blade	1	A	40		60		25	40	9	170-230	180°R=T
	2	B	40	62000	60	92000	25	40	9	170-230	180°R=T
Disc	1	C	28		54		20	30	1.5		150°R=6mm
	2	A	28	43000	54	83000	20	30	1.5		150°R=6mm
	3	D	75	116000	90	140000	15	40	6	260	
	4	B	50	77000	70	109000	16	35	5		
Pinion	1	A	50	77000	70	109000	20	40	5	2000	
	2	D	75	116000	90	140000	15	40	6	260	
	3	C	70	109000	90	140000	15	45	Charpy 8		
Rims		A	54-60	83000-93000	T.S. x 1.5 elong			90			180°R=16mm
Gear Wheels for Auxiliaries			31-48	48000-74000	52-68	80000	25	40		135-196	
Turbine	1	B			45-60		22		4		120°R=25mm
Cylinders	1					70000-92000					
	2	A			45-57	70000-88000	15				90°R=25mm
Nozzles	1	A	28	43000	48	74000	25			160	180°R=T
	2	C	28	43000	45	70000	25			160	180°R=T
Cast Iron	1				23	35000					
	2				14	22000					

ENCLOSURE (B)

REPORT ON REDUCTION GEARS
Tests for Pitting and Tooth Loading

An experiment was undertaken between April 1943 and June 1944 to determine what increase in tooth pressure and pitch line speed might be permitted over those then in use. (See NavTechJap Documents No. ND50-1900, 1902 and 1903, Enclosure (D)).

As the documents indicate, the later stages of the experiment were devoted to study of the substitute materials that had to be used due to the need to conserve the supplies of nickel.

An air raid in May 1945 practically destroyed the data that had been assembled from the test so that much of the information obtained is from the memory of those engaged in the experiment.

The test set up is the conventional"back-to-back" torque test, common in the U.S.A. It is of interest to observe from the Japanese comments that those engaged in the experiments were under the impression that a radically new scheme of gear testing had been developed.

Apparently the standards established for Navy gears contemplated a maximum tooth load of 1000 pounds per inch of face and a top pitch line speed of 150 ft/s. The tooth pressure could be increased 30% if the pitch line speed remained unchanged but, if the speed were increased 20%, the load was not to go up by more than 20%.

One can only assume that the test data had not been adequately evaluated as all of our experience is definite that the higher speed elements of a double reduction gear exhibit a much superior load supporting characteristic in respect to pitting than is the case with the lower speed second reduction gears. The tests currently underway at the Philadelphia Navy Yard give support to this prior experience.

The gears which the Japanese tested had the following dimensions:

<u>Single Reduction</u>

P.C.D. pinion	12.426"
P.C.D. wheel	138.187"
No. teeth, pinion	58
No. teeth, wheel	645
Angle of helix	30°
D.P.	5.3 (Circular pitch 0.673')
Ratio of reduction	11.12

The maximum speed attained during the test was 3336 RPM of the pinion for a period of 10 hours. This is equal to a pitch line speed of 180 ft/s. According to the chart included with the documents referred to above, a tooth load of 1460 lbs. per inch of face was carried. This is equal to a K factor of 129.

Conventional roller tests made to study the phenomena of pitting present nothing new. The work of Stewart Way in Pittsburg some ten or more years ago produced similar information. It may be that when the Japanese documents are translated completely other information will be disclosed, but the present summary of the documents gives no outstanding data and, in agreement with previous investigators of the subject, regards pitting as unavoidable unless the loads to be carried by the teeth are reduced to what would be an uneconomical level.

ENCLOSURE (C)

REPORT ON VIBRATION OF TURBINE BLADES
by H. TOHARA

A. History of Experiments

These experiments covered a period from 1937 to 1942. Formerly it was supposed that the breaking of turbine blades was caused by resonant vibration of the first order.

Figure 1(C)

RESONANT VIBRATIONS

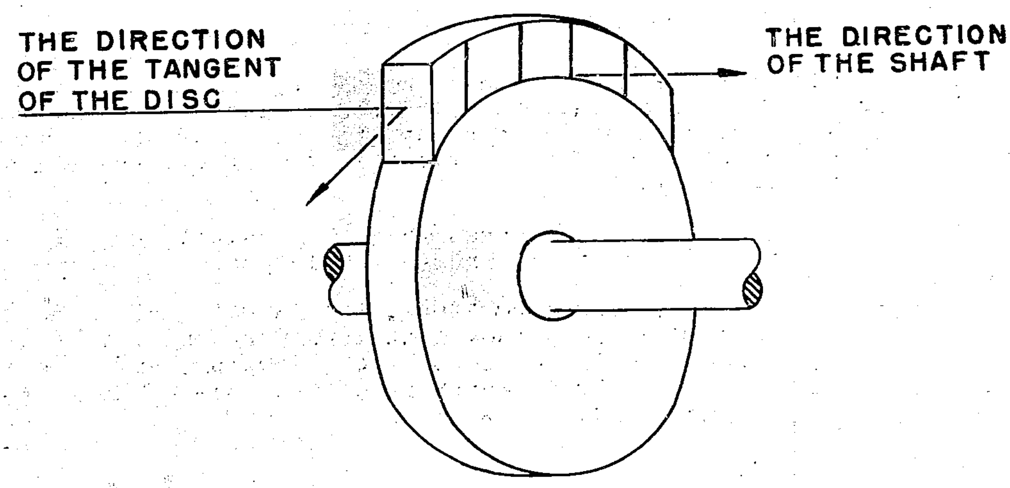

Figure 2(C)
DIRECTION OF VIBRATION

On this assumption the maximum stress would occur at the clamped end or root of the blade. Actually however, breaks frequently occurred near the middle of the length, between one third of the length from the top and the middle point.

A turbine blade vibrates tangentially with respect to the disc on which it is mounted as well as in the direction of the shaft. The former is the principal cause of breakage. The tests therefore took into consideration the resonant bending vibration of the higher order as a cause for the breaking of the blades.

Experiments were made first on actual and model single blades and then on groups of such blades. Actual blades and model blades which had a uniform rectangular cross-section were secured to an iron block weighing 200 kilograms.

ENCLOSURE (C), continued

B. <u>Single Blades</u>

The arrangement of experimental apparatus is shown in Fig. 3(C).

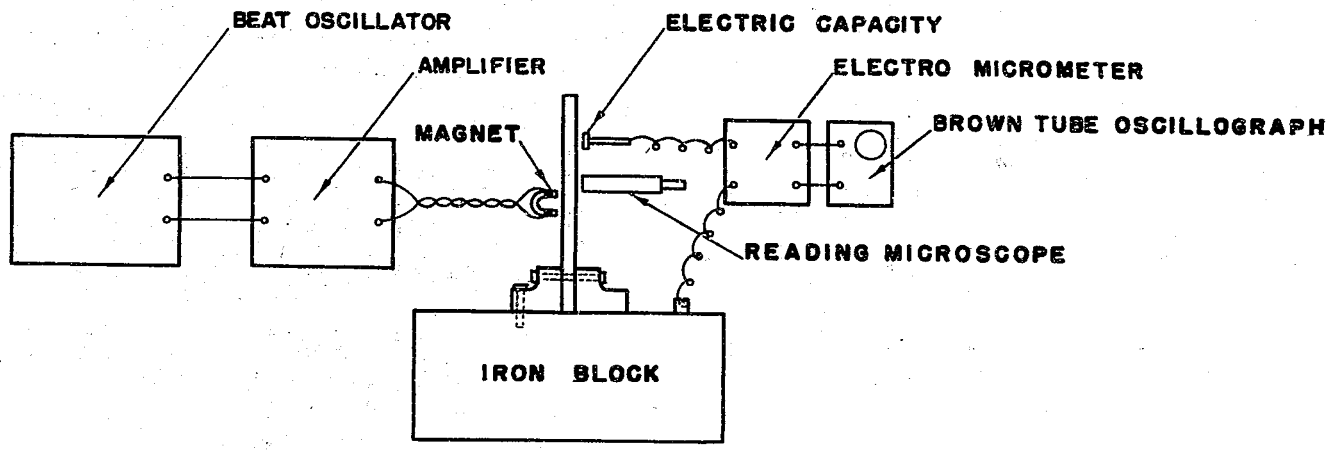

Figure 3(C)
ARRANGEMENT OF EXPERIMENTAL APPARATUS

The amplitude of blades was measured by means of electric capacity as well as a reading microscope. The vibrational form of actual blades is similar to that of model blades in the case of the resonant vibration of the first order, but in the case of the resonant vibration of the second order, the former has a higher nodal point than the latter.

The damping curve of the actual blades shows that they damped more rapidly than was expected from the data of their internal friction.

The logarithmic decrement of 13% chromium steel, the material of these turbine blades, is smaller than 2×10^{-3}, when they damp merely by their internal friction. The actual blades used in the test block, however, have the logarithmic decrement larger than 5×10^{-2}, which shows that the greatest friction of blades lies in their clamped end.

C. <u>Group of Turbine Blades</u>

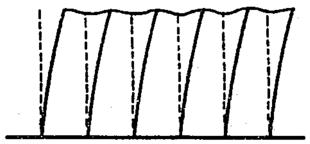

Figure 4(C)
RESONANT VIBRATION CURVES OF THE FIRST ORDER

ENCLOSURE (C), continued

SYMMETRIC ANTI-SYMMETRIC

Figure 5(C)

RESONANT VIBRATION CURVES OF THE SECOND ORDER

Singularity occurs in the case of the group of turbine blades when in the state of resonant vibration of the second order. There are symmetric and anti-symmetric forms. Consequently, there are many frequencies of resonant vibration of the second order.

Shrouding also influences the natural frequencies of a group of turbine blades, but it is difficult to calculate the natural frequencies and consequently they must be obtained by experiment.

D. Conclusions

A crisis occurs when some of the natural frequencies coincide with p x N, where p is the total nozzle number and N is the number of revolutions. When the maximum stress of the blade exceeds its fatigue limit, the generation of a crack is to be expected.

In most cases the critical frequencies are those of the second order.

ENCLOSURE (D)

LIST OF DOCUMENTS FORWARDED THROUGH ATIS TO THE BUREAU OF SHIPS

NavTechJap No.	ATIS No.	Title	Drawing No.
		TURBINE DRAWINGS FOR DD AKITSUKI	
ND50-1900	4610	Second Report on Pitting of Gear Teeth	
ND50-1901	4611	First Report on Pitting of Gear Teeth	
ND50-1902	4612	Test of propeller-type circulating pump	
ND50-1903	4613	Research Report on Pitting of Gear Teeth	
ND50-1904.1	4614	Cruising LP assembly	E130 L-1
.2		Cruising LP plan view	E130 L-2
.3		Main HP	E110 M-1
.4		Main IP	E110 M-1
.5		Main IP plan view	110 M-2
.6		Main LP plan	110 L-1
.7		Main LP plan view	110 L-2
.8		Main LP end view	110 L-3
.9		HP nozzle block	E111 H-8
.10		HP labyrinth sealing strips	E111 H-13
.11		HP 2nd and 3rd stage pinned blades	E112 H-5
.12		HP blade shrouds	E112 H-10
.13		Expansion joint in HP to IP crossover	E117 H-4
.14		Labyrinth gland HP turbine	E131 H-15
.15		Thrust bearing for cruising HP	E135 H-4
		TURBINE DRAWINGS FOR BB YAMATO	
ND50-1905.1	4620	HP turbine assembly	E110-4
.2		HP turbine assembly plan view	E110-5
.3		LP turbine assembly elevation	E110-6
.4		LP turbine assembly plan view	E110-7
.5		LP turbine assembly end view	E110-8
.6		Last row blades LP	E112-49
.7		Shroud bands LP	E112-30
.8		Expansion joint for crossover HP to LP	E117-3
.9		Turbine shaft direction indicator	E117-6
		TURBINE AND GEAR DRAWINGS FOR CV KATSURAGI	
ND50-1906.1	4615	Cruising HP assembly	E130-1
.2		Cruising LP assembly	E130-2
.3		Cruising LP assembly plan view	E130-3
.4		Main HP assembly	E110-7
.5		Main LP outside view	E117-25
.6		Main LP assembly	E110-2
.7		Main LP assembly plan view	E110-2
.8		Main LP assembly end	E110-3
.9		Main LP assembly cross section	E110-8
.10		Main LP assembly plan view	E110-9
.11		Main LP assembly end view	E110-10
.12		Main LP assembly outside view	E117-27
.13		Cruising reduction gear	E140-1
.14		Main reduction gear	E120-1
.15		Main reduction external fittings	E125-11
.16		Main reduction spray pipes	E125-12
.17		Design specifications	E911-9

ENCLOSURE (D), continued

REDUCTION GEARS FOR DD AKITSUKI

ND50-1907.1	4616	Gear case vent	E117-14
.2		Assembly main gear	120-1
.3		Assembly cross sections	120-2
.4		Machinery drawing upper case	121-1
.5		Fabrication drawing upper case	121-2
.6		Bearing brackets	121-3
.7		Plates for upper case	121-4
.8		Lower gear case	121-5
.9		Lower gear case details	121-6
.10		Lower gear case details	E121-1
.11		Upper gear case details	E121-2
.12		Upper gear case details	E121-3
.13		Upper gear case details	E121-4
.14		Bearing caps	E121-5
.15		Upper covers	E121-6
.16		Studs and bolts	E121-7
.17		Main gear bearings	E121-8
.18		Pinion bearings	E121-9
.19		Pinion bearings	E121-10
.20		Pinion bearings	E121-11
.21		Pinion bearings caps	E121-12
.22		Pinion oil rings	E121-13
.23		Oil pan	E121-15
.24		Foundation chocks	E121-16
.25		Bull gear assembly #344	122-1
.26		Bull gear assembly #365	E122-1
.27		Bull gear outline	E122-4
.28		Pinions	123-1
.29		Pinions	E123-1
.30		Material list	E123-2
.31		Oil piping	E125-1
.32		Oil piping	E125-2
.33		Oil piping	E125-3
.34		Clutch between cruising gear & IP	E144-3
.35		Design specification	

270 mm STANDARD AUXILIARY DRIVE

ND50-1908.1	4617	Assembly	3192-1
.2		Cylinder	-2
.3		Gear case	-3
.4		Cylinder cover	-5
.5		Throttle valve body	-6
.6		Rotor	-7
.7		Blade grooves	-8
.8		Nozzle block	-9
.9		Rotating blades	-10
.10		Stationary blades	-11
.11		Oil rings	-12
.12		Glands	-13
.13		Bearings	-14
.14		Auto stop throttle	-15
.15		Nozzle control valve	-16
.16		Sentinel valve	-17
.17		Bolts	-18
.18		Casing for overspeed trip	-19
.19		Weight for overspeed trip	-20
.20		Pressure gauge fittings	-21
.21		Piping	-22

ENCLOSURE (D), continued

AUXILIARY TURBINE AND GEAR ASSEMBLIES

ND50-1909.1	4618	120 mm for Condensate pump	E182-11
.2		180 mm for Fuel oil pump	E3113-1
.3		180 mm for Condensate pump	E184-5
.4		270 mm for Circulating pump	3238-1
.5		270 mm for Blower	3271-1
.6		270 mm for Feed	E311-5
.7		360 mm for Test unit	M8-4209
.8		360 mm for Feed pump	M8-14500

AUXILIARIES FOR DD ISOKAZE

ND50-1910.1	4619	Main oil cooler	T8494
.2		Main feed pump	T20010
.3		Circulating pump	39110
.4		Air ejectors	C1611
.5		Condensate pump	T19787
.6		Blower	L840-1